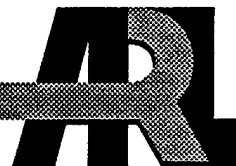


ARMY RESEARCH LABORATORY



Vibration Diagnostics for the M1 Abrams Tank Gas Turbine Engine

Thomas A. Korjack

ARL-TR-805

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1. INTRODUCTION

One of the primary functions of turbine engine maintenance programs is to reduce the possibility of mechanical failure by implementing effective preventive measures, without incurring excessive cost or equipment downtime. Most of the measurements presently made to determine engine health are engine temperatures, pressures, and speeds. These measurements provide a complete picture of the gas path characteristics of an engine. The more difficult and, therefore, the less often made, measurements are those related to the mechanical integrity of the engine. While it is true that mechanical deficiencies affect engine performance, more often than not, the performance change noted is an engine stoppage (Frarey and Handjani 1966).

The following measurements are usually made during postoverhaul testing and subsequent operational phases of the gas turbine engine:

- (1) Shaft horsepower or thrust
- (2) Exhaust gas temperature
- (3) Revolutions per minute (rpm)
- (4) Exhaust area
- (5) Fuel flow
- (6) Oil pressure
- (7) Run-down time
- (8) Vibration (mils, double amplitude).

Measurements 1-5 are essentially recorded to determine if the engine is performing to specifications. Measurement 6 is primarily a damage prevention measurement.

If pressures do not meet test specification, the engine will be shut down. Measurement 7 checks inadequate rotor/shroud clearances and general engine alignment tolerance. Providing these initial measurements meet specifications, measurement 8, vibration, is almost wholly depended upon as an indicator of engine mechanical integrity. It must indicate faults, such as:

- (1) Unbalance
- (2) Gear box condition

- (3) Bearing condition
- (4) Accessory condition
- (5) Mechanical alignment
- (6) Installation procedures.

There is no question that engine vibration is influenced by each of the foregoing conditions; however, the vibration sensitivity to each condition varies widely. A qualitative plot of engine vibration introduced by each of three conditions of varying seriousness may be presented as shown in Figure 1. It can be seen that a rather serious bevel gear box condition will or will not be detected, depending upon the degree of compressor unbalance that may exist as a bias. In addition, it is evident that for an otherwise healthy engine, the condition of the fuel pump would have to be dangerously close to disintegration before its vibration would be cause for engine rejection (Curtiss-Wright Corporation 1968; Frarey and Zabriskie 1965; and Mechanical Failures Prevention Group 1971).

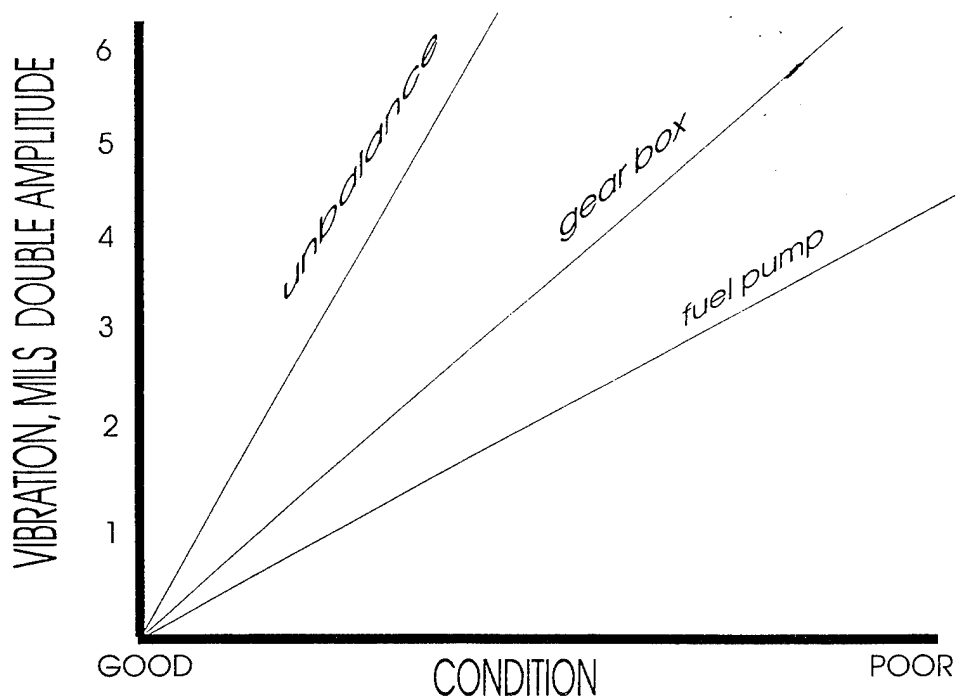


Figure 1. Engine vibration sensitivity to various malfunctions.

The problem of establishing the mechanical condition of components is even more difficult during an engine's operational life since accessibility, support equipment, and skill levels are less available in the field than at an overhaul depot. Even skilled mechanics cannot perform adequate early diagnosis of incipient failures using presently available tools. What is clearly needed is a method of establishing confidence in the mechanical integrity of an engine during both operational use and postoverhaul testing.

A method of monitoring turbine engine operation for incipient malfunction, designated the "diagnostic sonic analyzer technique," has been developed (Frarey and Handjani 1966). The sonic analyzer and a related technology is a preventive maintenance tool that uses the acoustical radiated energy produced by the rotating components to measure the operating condition of gas turbine engines and associated power train systems. Using this tool, there is no need to even partially disassemble the installation, nor physically attach any instrumentation to the equipment being monitored.

The sonic analysis technique has been used since 1963 to "listen" to gas turbine engines and drive systems. Since 1969, the United States and other military services have employed it as an effective maintenance tool. As an outgrowth of this experience, the sonic analyzer is now being applied to detect the mechanical condition of industrial gas turbine installations and to assist in development testing of planetary gear boxes and transmissions. The analyzer is adaptable to any gas turbine or transmission system by use of applicable engine-transmission plug-in modules and programming.

The sonic analysis concept may be considered an outgrowth of vibration analysis. Vibration analyzers have been developed that detect component mechanical degradation by producing a vibration spectrograph of the engine. Discrete frequencies observed in the spectrograph could be related to individual components by a knowledge of the mechanical details and engine speed.

Initial investigations on gas turbine engines and related drive systems showed that the same type of spectrograph could be obtained by analyzing the output of microphones placed in the extreme near field of the engine. It was found that all discrete frequencies detected in the acoustic spectrum could be related to individual engine components. The use of microphones to supplement vibration transducers also provided additional benefits. An increase in frequency response was obtained and indicated that the meaningful spectrum often exceeded 20,000 Hz. By using omnidirectional microphones, only two or three microphones are required to detect all mechanically generated frequencies of interest produced by the gas turbine engine.

The sonic analysis concept is based on the analysis of discrete frequencies emitted by individual rotating components of gas turbine engines and associated power train systems. It utilizes microphones placed in the extreme near field of the engine for sensing these frequencies. The computation of these frequencies depends upon such factors as the number of teeth on mating gears, the number of blades on compressor and turbine rotors, the number of rolling elements and dimensions of bearings, and the shaft revolutions per second (rps) of rotors, gears, and bearings. Narrow band filtering is employed to separate the discrete frequencies from the noise components to determine the signal characteristics relative to amplitude, harmonic, and modulation content.

When a part or component begins to wear or go through some other physical change, the character of the acoustic signal is altered. By monitoring the characteristics of all the discrete frequencies produced by the rotating components, it is possible to detect changes in mechanical condition and to pinpoint the individual component that is deteriorating.

The total spectrum produced by any mechanism may be said to have two components:

- (1) The noise spectrum
- (2) The line spectrum or the family of discrete frequencies produced.

The noise spectrum, although it undoubtedly contains information on the source condition, is extremely complex and almost impossible to separate.

Each of the discrete frequencies would be produced by one component. If the engine were a simple rotating compressor, for instance, one frequency might be produced by a gear train, one by rotating compressor blades, and one, two, or three frequencies produced by ball or roller bearings. In most engines, the frequencies produced by different components are not related other than by the fact that they are all functions of shaft rotation. This causes one to look for signals from a gear train at one point in the spectrum and signals for an oil pump at another point.

These spectrograms would merely be academic interest if it was not possible to show there is a definite relationship between the character of the sounds and the mechanical condition of the machine. Laboratory tests on gear trains, bearings, and compressor stages run with varying amounts of inflicted damage resulted in corresponding signal changes in the sound (Curtiss-Wright Corporation 1968; Frarey

and Handjani 1966; Frarey and Zabriskie 1965; Mechanical Failures Prevention Group 1971). While these tests were not exhaustive of all amounts and kinds of defects, they were indicative of the kinds of changes to expect as a component of the machine deteriorates.

Analysis of vibration and sonic data has shown that there are three main signal characteristics related to mechanical degradation. These are:

(1) Amplitude: Simple amplitude changes are the most common and easiest to detect. However, all discrete amplitudes alone cannot be used to reliably determine engine condition since they also change with rpm and load. Large variations in amplitude are useful indications, since they are indicative of drastic changes to the component.

(2) Modulation side bands: The most useful tool in malfunction detection is the determination of amplitude uniformity of a component signal. If the gear train is operating properly, an impact sound is made each time a tooth comes in contact with a tooth in the adjacent gear. As each successive tooth contacts the next gear, an identical pulse is generated. The number of these pulses generated per second is the characteristic tooth contact frequency and is equal to the product of the number of teeth on the gear and the rps of the gear. Also, the presence of uneven tooth wear, broken teeth, etc., will produce higher amplitude impact sounds, which produces new frequencies on either side of the original discrete frequency. These new frequencies are called side bands, and the pure tone is now said to be modulated. Generally, the amplitude nonuniformity is repeated every gear revolution, in which case the side bands are displaced from the center frequency by a value equal to the rps of the gear.

The indicator for gear condition now becomes the ratio of the side band amplitude to center frequency amplitude. This is a reliable predictor because the ratio only changes by 5–10% when the amplitude of the tooth contact signal increases by as much as 100% due to load.

(3) Harmonic content: Any wave that has other than a sine wave shape contains harmonics of the fundamental frequency. The number and relative amplitude of the harmonics depends on the shape of the pulse. The shape of the pulse, in turn, depends on the characteristics of the component producing the pulse. Theoretical studies and laboratory tests with data from turbine engines have shown that the relative amplitude of a set of harmonics is a good indicator of component wear. A change from sawtooth to square pulse is an example of this kind of signal change. It is the relative amplitude of higher harmonics

to the fundamental frequency (not absolute amplitude) that determines the pulse shape. As such, the pulse shape is not as sensitive to load changes.

Use of the signal analysis concept has been successful in diagnosing the following types of incipient failures or malfunctions:

- Compressor: Foreign object damage and rotor unbalance or eccentricities.
- Bearings: Rolling element defects, inner and outer race defects, and detrimental effects of oil starvation.
- Pumps: Fuel pump, oil pump, and hydraulic pump discrepancies.
- Gear box: Misalignment, improper mounting, improper mounting of accessories and gear, and bearing malfunction.
- Gears: Damaged teeth, gear tooth scuffing, and gear eccentricities.

2. CONCEPT

A diagnostic system should be developed that demonstrates the application of the sonic concept and analysis techniques to monitoring the mechanical degradation of turbine engines and associated power train systems. Inputs from a number of sensors should be normalized to the level of a predetermined band of noise that is characteristic of a given engine, thereby providing a reference base for component condition limits and analyzer system calibration.

Besides the data input, one or more of the microphones should monitor the engine rpm by sending a discrete signal produced by an engine component into a phase locked filter. The output of the phase locked filter should then be processed by a frequency ratio generator. The resulting engine rpm tracking function enables precise control of the narrow band center frequency placement.

The location and number of microphones required for complete signal coverage of the sound spectrum should be determined during a preliminary data-acquisition survey. This survey should also be performed

to determine the component signal(s) to be utilized for rpm tracking and to establish basic component limits for the diagnostic program. Further refinement of the diagnostic programs is accomplished during training and implementation of the system.

A real time spectrum analyzer (RTA) generates the spectrum analysis of the vibration data instantaneously and in an easily comprehensible form. Rather than displaying a signal as a function of time (as an oscilloscope does), it displays the signal in terms of its frequency content.

Other techniques for data acquisition, such as Signature Ratio Analysis, a technique for adding clarity and continuity to rotating machinery data; spectrum averaging, which improves the signal-to-noise ratio in the analysis of noisy signals; and spectrum translation, which allows extremely fine resolution, are available to the user. The data management task can be greatly facilitated by using the real time analyzer in conjunction with a small digital computer. A micro-computer can be used to store baseline information from which behavioral trends of a machine can be monitored.

Cataloging of vibration and noise spectra taken at regular intervals on a piece of equipment can be extremely valuable, both in diagnosing potential problems and in predicting machine life. There are two approaches that can be taken in this type of program. First, when a large number of the pieces of equipment are available for testing, vibration data can be taken for a batch of these components. From this data, a mean criterion for vibration spectra can then be established. Individual items may then be tested against this mean value and decisions made on a Go/No-go basis. Secondly, as in the case of a gas turbine or other large machines, vibration data may be taken at the beginning of the machine's service life and at periodic intervals thereafter. Every machine will exhibit changes in its vibration spectra due to gradual wear. By monitoring the vibration on a regular basis, any irregular changes in vibration level can be spotted before they get to a point that could cause actual machine damage.

The overall vibration monitors are dedicated sentries working full time and are very much a necessary part of maintaining safe machinery operation. In fact, one very worthwhile program for implementing the use of a spectrum analyzer is in response to a first-stage alarm given by such overall monitors; then, by exploring the spectrum, accurate judgements can be made and, perhaps, an emergency shutdown prevented. As an example, perhaps a decision can be made to decrease the load or speed of a machine by 10%, thereby extending its operation through an important production period until repair or replacement can be scheduled.

A real breakthrough for on-line monitoring was made available with the real time spectrum analyzer. It allows a spectral picture of a signal to be seen instantaneously. The basic limitation in using a swept frequency narrowband filter for analysis is the length of time required for the analysis. The narrower the filter, the more slowly it must be swept through the analysis frequency range.

The RTA accomplishes an analysis in a short period of time. This is primarily due to the technique of digital sampling and time compression that speeds up the input data signal. This signal is translated to frequencies in the megahertz range. Analysis of this proportionately higher frequency can be done rapidly using a wide band filter. Relative to the input signal frequency range, this filter is narrow.

An averager is a very important part of the RTA system. In cases where the vibration signal contains noise, the important vibration components can be buried or concealed by the noise. The averaging process allows the randomly occurring signals to approach a mean value, clearly revealing the important spectrum data.

A frequency translator allows for an ultra-narrowband filter to be used in the higher frequency range. This capability is important if vibration components are close in frequency. A typical example of this is where sidebands are generated due to an amplitude-modulation process (Fieldhouse 1970). The RTA can "zoom in" to separate and identify discrete signals when used with a translator.

One problem associated with vibration spectrum analysis can exist if rpm fluctuates slightly during the course of measurements because of inherent speed control limitations or, perhaps, because of fouling. When this occurs, different signature analysis records of a given machine will show vibration components occurring at different frequencies in the spectrum. A slight change in operating speed will produce an increasingly larger change at higher multiple order frequencies. A solution to this problem lies in the use of a Signature Ratio Adapter with the RTA.

Signature Ratio records data directly in vibration amplitude vs. ratio (of component to fundamental speed) rather than vibration amplitude vs. frequency. In essence, it divides the frequency axis of the spectrum plot by rotational frequency of the machine. The harmonic relationships are automatically maintained at the same value regardless of speed changes that occur between the data-taking periods. The signature ratio concept also allows for order-related components to be more easily compared when looking

at records produced at two specifically different running speeds without having to do any mathematical gymnastics.

Any smaller speed variation occurring during the course of analysis, while supposedly operating at fixed speed, are automatically compensated for by using this technique. Thus, analyses performed in terms of signature ratio may be directly related to one another without mathematical correction. More importantly, suppose that spectrum averaging is used to deemphasize the background noise and to permit critical higher order vibration components to be visible. Without signature ratio, any slight speed change that occurs during the averaging process will cause the higher order components to change frequency, and a smearing effect will result. Averaging using the signature ratio concept can be accomplished without concern over minor speed variations.

The computer can do the otherwise manual task of predicting trends or of diagnosing machinery characteristics with a high degree of accuracy and at a very rapid rate. This can be done by giving the computer a machine's baseline performance spectrum data along with data gathered at regular intervals. The baseline information is gathered during installation of a new machine or following the overhaul of existing machinery. In the case of a machine that has been running for an extended period, useful baseline information can be established as long as performance and vibration levels have been steady.

The second role in which a computer is useful is in a completely automated system. Many such systems are currently in use providing early warning of incipient problems of failures. These systems generally monitor temperatures, pressures, flows, and overall vibration data. They provide different levels of warning based on trends that are observed from these inputs (Gas Turbine World 1971). Some systems even have diagnostic capabilities. Few of these systems use vibration spectrum data as inputs. This is because vibration spectrum information prior to the advent of the real time analyzer was impractical and costly. Today, the relatively low cost of the RTA makes this concept economically possible. Monitoring the vibration spectrum data in lieu of the overall vibration data yields a clearer picture of the machine's condition.

Displacement transducers, velocity transducers, accelerometers, microphones, and strain gages each provide vibration and noise information useful for this type of data acquisition. Each transducer has specific operational characteristics that would limit a good data acquisition program if only one type of transducer were used.

The displacement transducer, specifically the noncontacting probe, is a unique transducer in that it can measure the relative motion between two surfaces. It is the only transducer that can provide this type of information. Velocity and acceleration transducers measure forces that are relative to the earth's gravitational force. The displacement transducer has a good linear amplitude range, but is limited to operation in the lower frequency ranges. Nicks and scratches on a rotating surface directly affect the output signal and spectrum characteristics when using a displacement transducer.

The velocity transducer operates in a frequency range that overlaps that of the displacement transducer. The velocity transducer should not be used at low frequencies where its amplitude response becomes nonlinear due to resonances. It has a limit in its upper operating frequency range, but it extends beyond that of the displacement transducer. By applying the velocity transducer output to an electronic signal integrator, a displacement level can be read. The velocity transducer is useful to determine vibration that exists at higher orders of rotational speed where a displacement transducer cannot respond. Many structural resonances occur in this frequency range. The velocity transducer, therefore, is useful to determine these parameters of a machine's behavior.

There are many characteristics of a machine where extremely high-frequency information exists. For example, blade passage frequencies can exist at 100 times running speed. Gear mesh frequencies and, in some cases, bearing frequencies can produce vibration information at these higher frequencies also. For this type of data, an accelerometer is required due to its extremely high-operating frequency characteristics. The output of an accelerometer is calibrated in units of gravitational forces called "g's." Acceleration is the second derivative of displacement, which means that if its signal is passed through an electronic double integrator, the output can also be read in displacement units. The formula, $g = 0.0511f^2x$ (where x is peak-to-peak displacement and f is frequency in revolutions/s), is derived from this relationship and provides a "connection" between the two units of measurement. Another advantage in using an accelerometer is the fact that it is relatively light and may not add appreciable damping to the structure.

In many cases, a microphone is useful for surveying running machinery. The narrowband acoustic spectrum produced by the RTA allows accurate identification of the frequency at which noise exists. Knowing the precise frequency information, the source of the noise or vibration can be identified. Where the microphone allows for taking a "broad view survey" to pinpoint a possible problem, it does not lend itself to repeatability. Since it is not a directional transducer, the transmission paths for the noise cannot

be defined. Many reflected transmission paths can be severely influenced by surroundings that might change from one time to another.

The strain gage cannot be overlooked as a valuable transducer when study is being made of the mechanical integrity of a machine. Spectrum data from the strain gage can be equally important to the design engineer as is the value of strain itself. Often a strain gage can be placed in areas on a structure where it is impossible to use other types of transducers due to a lack of space.

Much information stands to be gained by simply using a spectrum analyzer when a problem is suspected. In this case, it may not be necessary to compare to the spectrum with previously obtained data. Many excellent references are available that give guidelines for correlating basic machinery problems with their vibration spectra (Shuey 1972; Tustin 1971; Miller 1967). The RTA has the capability to provide the spectrum information, instantaneously revealing a machine's characteristics. A complete listing of all possible frequencies at which vibration could exist should be compiled and kept with the machine's portfolio. Such a listing should include information like running speed, types of bearings, number of elements and size (if roller bearings are used), gear mesh frequencies, number of turbine blades, speeds of accessory components, and information about structural resonances or critical frequencies. This would provide a rapid means by which the spectrum data could be resolved and frequency components where high vibration levels exist could be identified. Soon a portfolio of case histories and vibration trend data would be compiled and used to make accurate predictions concerning the machine's operational characteristics and life expectancy.

A dedicated program of spectrum analysis would certainly be more fruitful. The spectrum of a signal from a vibration transducer produced by a running machine can be used as the "signature" or fingerprint of that particular machine's operational condition (Bannister and Donato 1971). The signature data should be carefully labeled with a date, pertinent operating conditions, transducer type and location information, and filed in a folder assigned to the particular machine.

A program of periodic spectrum analysis would provide new signatures to be compared with the baseline information. Any rapid changes with time in the signature could be carefully observed by doing signature analysis more frequently to keep track of any trends that might exist. Normal wear over long operating periods should show a gradual trend that would be reflected in the analyses. If a particular frequency component begins to change at an increasing rate, it should be looked upon as an indicator of

impending problems. This new trend in the spectrum data can be compared with trends in temperature and other data to help identify the nature of the problem. Accordingly, an orderly shutdown may be scheduled.

3. RESULTS AND CONCLUSIONS

Spectrum and sonic analysis techniques have been developed and successfully applied as maintenance tools used for monitoring the mechanical condition of rotating components in gas turbine engines and drive systems operating in aircraft and test cell installations as evidenced in Curtiss-Wright Corporation (1968), Frarey and Handjani (1966), Frarey and Zabriskie (1965), and Mechanical Failures Prevention Group (1971). The concept has been applied to determine the mechanical integrity of industrial gas turbine installations and to monitor the dynamic operating characteristics of transmissions during development testing.

Diagnostic monitoring of a transmission system achieved excellent correlation between acoustic and spectrometric oil analysis techniques in the case of a spalled roller on the left nacelle pinion bearing (Curtiss-Wright Corporation 1968; Frarey and Handjani 1966; Frarey and Zabriskie 1965; Mechanical Failures Prevention Group 1971).

Spectrum and sonic analysis techniques have been implemented as standard overhaul maintenance procedures to assure a higher degree of quality control during post overhaul testing of customer gas turbines. This added capability has also reduced maintenance costs related to establishing engine acceptance by defining the level of maintenance required for corrective action.

Use during production overhaul testing has demonstrated the capability to predict the amount of unbalance present in compressor rotors from the value of the sideband amplitude ratios. Engines rejected for excessive vibration, but found to be mechanically sound by the sonic engine analyzer, have been further investigated for faulty vibration transducers to prevent possible rejection back to overhaul.

Data have been recorded and analyzed for uprated power development testing of planetary gear boxes of double helical gear configurations. The signal characteristics detected in the acoustic spectrum were an indication of mechanical and dynamic changes occurring during the various testing phases. They were related to carrier eccentricity, unequal planet loading, and degree of applied torque. These data, coupled

with that obtained from other methods of vibration instrumentation, have been correlated with results derived from mathematical models to assist in defining a specific direction for design and testing. Acoustic patterns obtained during locked-torque testing have been related to differences in system fits and clearances. The degree of amplitude modulation of gear-meshing frequencies and detection of resonant frequencies at various operating conditions were monitored to assist in defining test profiles for the experimental assemblies being evaluated.

As a trial run for a proof in concept principle, Fast Fourier Transforms (FFT's) were generated from both a turbine start-up condition and gearbox rundown of an M1 Tank Turbine engine. Data was collected on the engine via an accelerometer placed originally on the turbine bearing shaft housing and then moved to the gearbox housing for the coast down. Since we did not have the luxury of a tachometer signal, much of the data was difficult to reduce to each individual component; however, these data points/plots demonstrated that the two locations chosen were very signal rich and perhaps may give us a good start in determining the proper course of technical action when funding and equipment become available.

Pictured in Figure 2 is the time domain data from the turbine startup.

A "waterfall" data display can then be generated via Figure 3, which clearly shows some of the engine signal peaks paving the way for further research and investigation.

Selection of a couple of FFT's from Figure 3 were isolated to show how a signal can be decomposed via spectrum analyzer, as depicted in Figures 4 and 5. Figure 5 in particular is from the steady-state speed data towards the end of the data file collected.

Figures 6 and 7 show the data taken from the gear box alone. This location also indicated a signal rich environment for further research and investigation. Better locations certainly exist to take data samples, but the locations thus chosen proved that signal-rich analyses can be achieved via spectrum analysis for the M1 turbine engine to be used in engine diagnosis and performance.

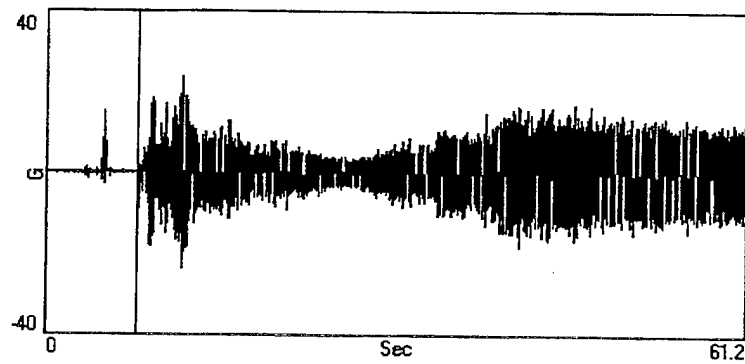


Figure 2. Time-domain distribution of frequency vs. time of a rotating center shaft for an M1 GAS turbine engine.

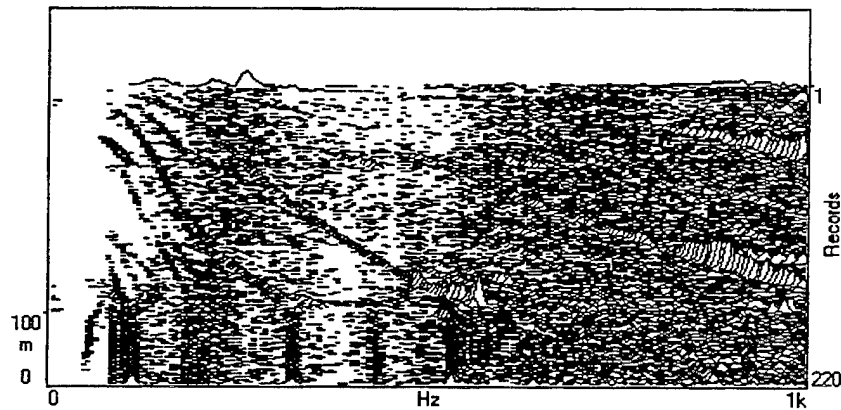


Figure 3. Time-domain distribution of frequency vs. time for a rotating center shaft (waterfall display) for an M1 GAS turbine engine.

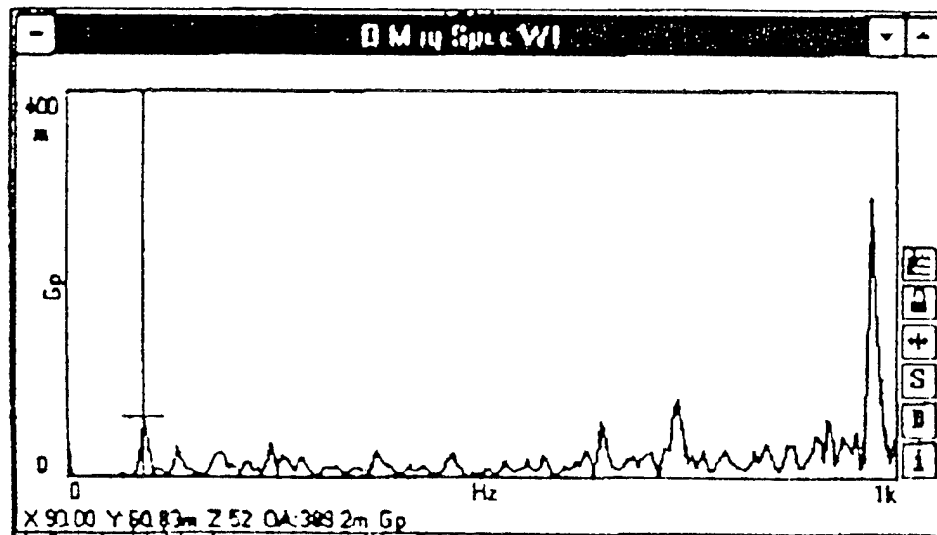


Figure 4. Decomposition of time distribution signal of a rotating shaft for an M1 GAS turbine engine.

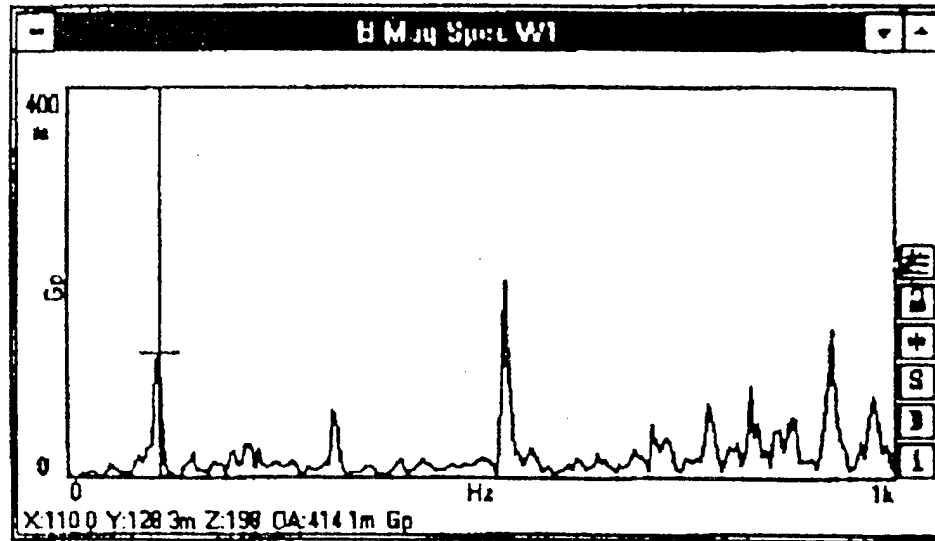


Figure 5. Decomposition of time distribution signal of a rotating shaft for an M1 GAS turbine engine.

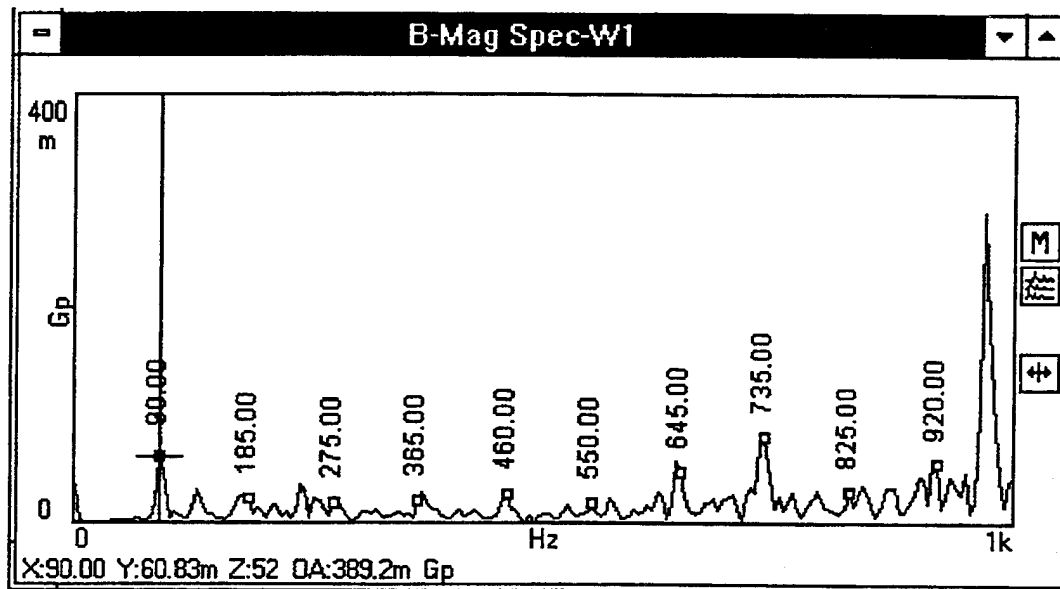


Figure 6. Amplitude vs. frequency signal of the gear box signal of an M1 GAS turbine engine.

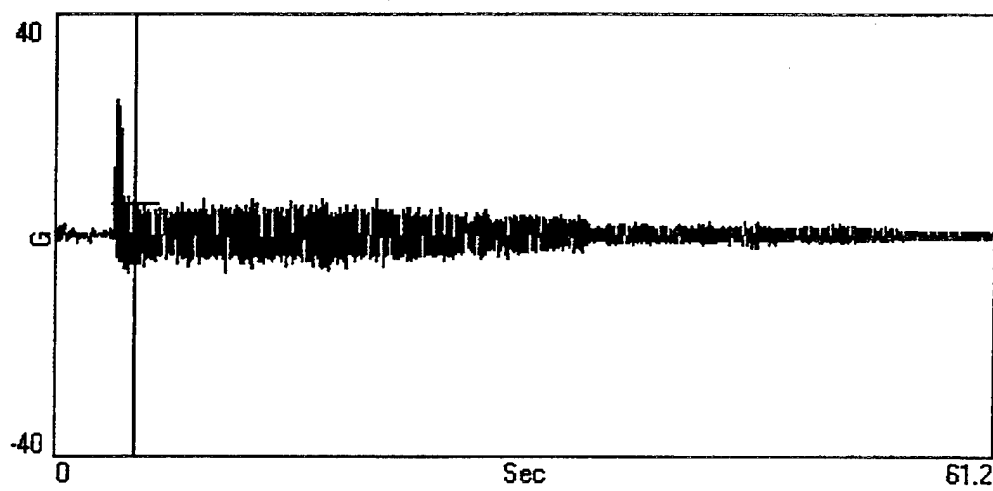


Figure 7. Frequency vs. time signal distribution of the gear box of an M1 GAS turbine engine.

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